NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TECHNICAL NOTE 2841

INVESTIGATION OF 75-MILLIMETER-BORE DEEP-GROOVE BALL BEARINGS UNDER RADIAL LOAD AT HIGH SPEEDS

I - OIL-FLOW STUDIES

By Zolton N. Nemeth, E. Fred Macks, and William J. Anderson

Lewis Flight Propulsion Laboratory
Cleveland, Ohio

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SUMMARY

Two 75-millimeter-bore (size 215) inner-race-riding cage-type ball bearings were used in an experimental investigation of the effects of load, oil flow, and oil inlet conditions on bearing operating characteristics over a range of DN values (bearing bore in mm times shaft speed in rpm) from 0.3×10^6 to 1.2×10^6, static radial loads from 7 to 1113 pounds, and oil flows from 2 to 8 pounds per minute; the absolute viscosity of the oil at an inlet temperature of 100° F was 42.6×10^{-7} reyns (34.5 centistokes).

The radial location of the oil jet and the distribution of the oil were found to be important factors in the lubricating and cooling effectiveness of a given quantity of oil. Lowest average bearing temperatures were obtained when the oil was directed at the space between the cage and the inner race.

The quantity of the oil which flowed through the bearing had an important effect on bearing operating temperatures and lubrication-system heat load. Outer-race temperature, which was found to be a function of the quantity of oil transmitted through the bearing regardless of the lubrication method, decreased with increasing transmitted oil flow. At a given total oil flow, the outer-race temperature was a minimum when all the oil was made to flow through the bearing (achieved by puddling). The inner-race temperature was dependent on the lubrication method and was greater than the outer-race temperature when all the oil flowed through the bearing. At constant total oil flow for single-jet lubrication, the power rejected to the oil increased with increasing transmitted-oil flow and was maximum when all the oil flowed through the bearing.

INTRODUCTION

Ball bearings operate under severe conditions in turbojet-aircraft engines because of the high speeds, high loads, and high temperatures encountered. Very little information is available on the performance characteristics and the limitations of high-speed ball bearings or on how these characteristics and limitations are affected by variables such as load, oil flow, and oil inlet conditions. Some information is available on the effects of these variables on the performance and limitations of conventional cylindrical-roller bearings (references 1 to 4). Such information as is available on ball-bearing performance (references 5 to 8) is generally for DN values below 10^6. It is desirable to know the performance characteristics of ball bearings under radial load, because of the possibility of using ball bearings in turbojet-aircraft engines where roller-bearing operation has at times been marginal (reference 9).

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The object of this investigation was to study the performance characteristics of conventional ball bearings at high speeds under radial load. The effects of speed, load, oil flow, and oil inlet conditions on bearing operating temperatures are reported.

The ranges of controlled variables investigated for two ABEC 5 deep-groove ball bearings (size 215) were: DN values, 0.3×10^6 to 1.2×10^6; loads, 7 to 1113 pounds; oil flows, 2 to 8 pounds per minute; oil inlet temperature, 100° F; absolute viscosity at inlet temperature, 42.6×10^{-7} reyns (kinematic viscosity, 34.5 centistokes).

One oil was used in this investigation, and external heat was not applied to the bearing housing or to the shaft. This investigation was conducted at the NACA Lewis laboratory.

APPARATUS

Bearing rig. - The bearing rig (fig. 1) used in this investigation is the same as that used in the investigations reported in references 1 to 4. The bearing under investigation was mounted on one end of the test shaft, which was supported in a cantilever fashion for purposes of observing bearing component parts and lubricant flow during operation. A radial load was applied to the test bearing by means of a lever and dead-weight system in such a manner that the alinement of the outer race of the test bearing was essentially unaffected by small shaft deflections or by small shaft and load-arm misalinements.

The support bearings were lubricated in the manner described in reference 1. The oil was supplied to the support bearings at a pressure of 10 pounds per square inch through a 0.180-inch-diameter jet and at a temperature equal to that of the oil supplied to the test bearing (100° F).

The drive equipment is described in reference 1. The speed range of the shaft is 800 to 50,000 rpm.

Test bearing. - Two test bearings were used for this investigation although most of the data reported herein were obtained by use of only one of the bearings. These bearings were conventional aircraft-grade ball bearings (ABEC-5). The bearing dimensions were as follows: bore, 75 millimeters; outside diameter, 130 millimeters; and width, 25 millimeters. The bearings were equipped with a two-piece riveted retainer of laminated cloth-base phenolic material and eleven 11/16-inch-diameter balls (fig. 2). The retainer or cage was guided by the inner race. The operating conditions imposed on a cylindrical-roller bearing of the same size (215) in engine service are as follows: DN range, 0.3×10^{6} to 0.86×10^{6}; approximate gravity load, 375 pounds; and oil flow, 0.8 to 2 pounds per minute through a jet of 0.052-inch diameter.

Temperature measurements. - The method of temperature measurement is described in reference 1. Iron-constantan thermocouples were located at 60° intervals around the outer-race periphery at the axial center line of the bearing under investigation. A copper-constantan thermocouple was pressed against the bore of the inner-race at the axial midpoint of the test bearing; the voltage was transmitted from the rotating shaft by means of small slip rings located on the end of the test shaft (reference 10).

Lubrication system. - The general make-up of the lubrication system is described in reference 3. Changes were made to increase the upper limit of the oil pressure to the test bearing from 90 to 400 pounds per square inch. Oil shields were added to both sides of the test bearing so that the deflected oil and the transmitted oil could be collected and weighed. Oil flow to the test bearing was determined by calibrated rotameters.

PROCEDURE

Lubrication of test bearing. - Two methods of lubrication were investigated. In the first method, lubricant was supplied to the test bearing through a single jet having a 0.050-inch-diameter orifice. The jet was of type B described in reference 2, and had an orifice length-diameter ratio of 1. The oil was directed normal to the bearing face. In the second method (called puddling), the oil was introduced on one side of the bearing and forced to flow through the bearing to drain from the other side. 2672

One oil was used to lubricate the test bearings; the viscometric properties of this oil are given in figure 3. The oil was a commercially prepared blend of a highly refined paraffin base with a small percentage of polymer added to improve the viscosity index.

An oil sample was taken at the start of each oil run, at the start of each day's run, and at the conclusion of all the tests. Viscosities were obtained by standard laboratory procedures, and the data plotted in figure 3 represent the average for all the samples of each oil.

The oil was supplied to the test bearing at a temperature of 100° F, and at pressures of 20 to 400 pounds per square inch, which corresponded to oil flows of 2 to 8 pounds per minute. The variation in viscosity for the test oil as found in the laboratory tests was as follows:

Kinematic viscosity at 100° F (centistokes)			Maximum variation from mean (percent)
Minimum	Mean	Maximum	
29.96	34.5	40.20	17.82

Test bearing measurements. - The test bearing measurements were obtained in the manner described in reference 3 and are given in table I.

Reference conditions. - In order to determine the influence of running time on bearing performance, frequent checks of the bearing operating characteristics were made in the same manner as the checks on roller bearings described in reference 1. These checks were made at the following predetermined operating conditions: DN, 1.2×10^6 (16,000 rpm); load, 368 pounds; oil flow, 2.75 pounds per minute; jet diameter, 0.050 inch (in reference 1, the jet used had a diam. of 0.089 in.). These conditions hold for all curves except where noted. The oil samples taken during the runs indicated changes in the oil that might have accounted for changes in bearing operating characteristics.

RESULTS AND DISCUSSION

The results of the experimental investigation are presented in figures 4 to 17. As in references 1 to 4, bearing temperature was chosen as the principal criterion of operation, because it gives a good indication of the effects of all the operating variables.

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Effect of Ratio of Deflected-Oil Flow to Transmitted-Oil Flow on Bearing Operating Characteristics

<u>Effect of flow ratio on bearing operating temperatures</u>. - Early in this investigation, when the optimum radial location of the oil jet was being determined, it became apparent that the ratio of deflected-oil flow to transmitted-oil flow is an important variable regarding the effectiveness of ball-bearing lubrication. The marked effect of an increase in this ratio on bearing outer-race-maximum temperature for a constant total oil flow is shown in figure 4. The flow ratio was varied by varying the radial location of the oil jet. The inner-race temperature exhibited a somewhat erratic behavior with increase in the ratio of deflected-oil flow to transmitted-oil flow. Although a large portion of the oil passed through the bearing (low flow ratio) at jet-location e, where the oil jet is aimed at the space between the cage and the outer race, the inner-race temperature is high because most of the oil does not contact the inner race, but is flung outward by centrifugal action. The outer-race temperature is determined by the portion of the total flow which is transmitted through the bearing. The inner-race temperature is determined to a lesser extent by the portion of the total oil which is transmitted through the bearing.

It is obvious that more effective outer-race cooling was obtained with high transmitted-oil flows or low flow ratios.

Since radial location of the oil jet is critical regarding bearing operating temperature (particularly for an inner-race-riding cage) and since deflections and distortions occur in a jet engine as a result of maneuvering loads and thermal distortions, it is not expected that optimum lubrication can be maintained under all flight conditions. Special fan-tail jets may improve lubrication depending upon the magnitude of the deflections.

Effect of flow ratio on power rejected to oil. - Calculations were made to determine the effect of the ratio of deflected-oil flow to transmitted-oil flow on the power rejected to the oil by the method outlined in reference 4. The power rejected to the oil is maximum when all the oil is transmitted through the bearing (zero flow ratio) and decreases with increasing ratio of deflected-oil flow to transmitted-oil flow (fig. 5). Oil that flows through the bearing removes heat from the bearing; therefore, maximum outer-race temperature is a result of minimum power rejection to the oil.

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Effect of Independent Variables on Ratio of Deflected-Oil Flow to Transmitted-Oil Flow

Effect of DN. - The effect of DN on the ratio of deflected-oil flow to transmitted-oil flow is shown for bearings 19 and 20 in figure 6. The ratio of deflected oil to transmitted oil increases with increasing DN at a load of 368 pounds.

Effect of oil flow. - The effect of oil flow on the ratio of deflected oil to transmitted oil is shown in figure 7. In general, the ratio of deflected oil to transmitted oil varied from 1.2 to 2.4. The effect of DN on flow ratio is reduced at high oil flows; at oil flows higher than 4 pounds per minute, the flow ratio is nearly constant over the range of DN values investigated.

Effect of load. - The effect of load on the ratio of deflected oil to transmitted oil for bearing 19 is shown in figure 8. Over the range of DN values investigated, the ratio of deflected oil to transmitted oil decreases at loads of 7 to 113 pounds; at loads between 113 and 1113 pounds, the flow ratio increases with increasing load. The increase in the flow ratio with load may be a result of the shaft movement associated with an increase in load, which causes the impingement point of the oil to change with respect to the test bearing. Movement of the point of impingement of the oil stream toward the inner-race face would tend to increase the ratio of deflected oil to transmitted oil (fig. 9). This condition may not be representative of engine operation, but may be peculiar to the subject test rig.

Effect of radial location of oil jet. - The effect of the radial distance of the oil jet from the shaft center on the ratio of deflected-oil flow to transmitted-oil flow is shown in figure 9. The two lowest flow ratios were obtained when the jet was aimed at the spaces between the cage and races. When the jet was aimed at either of the races or at the cage, the ratio of flows was high; that is, little oil passed through the bearing.

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Effect of Independent Variables on Bearing Operating Temperatures for Single-Jet Lubrication

Effect of radial location of oil jet. - The effect of the radial distance of the oil jet from the shaft center on the operating temperature of bearing 20 is shown in figure 10. Oil was directed perpendicular to the bearing face at the six radial locations shown previously in figure 4. The data show that, for the six positions investigated, lower outer-race-maximum temperatures resulted when the oil was directed at the space between the cage and one of the bearing races. The inner-race temperature was lowest when the oil was directed at the space between the cage and the inner race (location c, fig. 4). For the lowest average bearing temperature, the oil should therefore be directed between the cage and the inner race.

Effect of DN. - The effect of DN on the outer-race-maximum and inner-race temperatures of bearings 19 and 20 is shown in figure 11. The outer-race-maximum temperatures increase with DN and are approximately linear functions of DN. The inner-race temperatures increase with DN at a rate greater than linear. It is evident that the difference in the operating temperatures of the two bearings is small for both the outer-race and the inner-race temperatures. These slight temperature differences are probably caused by differences in internal clearance and by slight variations in the flow ratio caused by mounting.

Effect of oil flow. - The effect of oil flow on the outer-race-maximum and the inner-race temperatures of bearing 19 is shown in figure 12, for DN values of 0.3×10^6, 0.735×10^6, 0.995×10^6, and 1.2×10^6, a load of 368 pounds, and an oil inlet temperature of 100° F. The outer-race-maximum and the inner-race bearing temperatures decreased with increasing oil flow; the rate of change of bearing temperature with oil flow also decreased with increasing oil flow. The decrease in temperature with an increase in oil flow occurs at a faster rate at the higher DN values.

Effect of load. - The effect of load on the outer-race-maximum and the inner-race temperatures of bearing 19 is shown in figure 13. At low DN values, load only slightly influences the outer-race-maximum and the inner-race temperatures of bearing 19 (fig. 13). At the higher DN values, the outer-race temperature increases appreciably with increasing load. The inner-race temperature also increases appreciably with an increase in load at the higher DN values except at loads below 113 pounds, where a decrease in load causes an increase in temperature. The increase in inner-race temperature may be caused by an increase in relative velocity between the cage and the inner race over that predicted from theory. This increase in velocity could have caused frictional heating of the inner race. Again, the quantitative results may be peculiar to the test rig and not representative of engine operation.

Effect of Independent Variables on Bearing Operating Temperatures for Puddling Lubrication

Effect of DN. - The effect of DN on the outer-race-maximum, the inner-race, and the oil outlet temperatures of bearing 20 is given in figure 14. The outer-race-maximum and the oil outlet temperatures are approximately linear functions of DN, and are very nearly the same at a given operating condition. The inner-race temperature is greater than the outer-race-maximum temperature at each operating condition. The differences in operating temperatures are more pronounced at the higher DN values. The inner-race temperature was lower than the outer-race-maximum temperature when the bearing was jet-lubricated (fig. 11).

Effect of oil flow. - The effect of oil flow on the outer-race-maximum, the inner-race, and the oil outlet temperatures of bearing 20 is shown in figure 15. Outer-race-maximum temperatures decrease with increasing flow and the rate of change of bearing temperature with oil flow decreases with increasing oil flow. Again, the outer-race-maximum and the oil outlet temperatures are very nearly the same. The inner-race temperature is greater than the outer-race-maximum temperature at each operating condition.

The inner-race temperature increases when the oil flow is increased from 2 to $2\frac{1}{2}$ pounds per minute; it then decreases with further increase in oil flow. The initial increase in inner-race temperature with oil flow is probably caused by the heating effect of churning oil in the bearing. A further increase in oil flow, although causing an increase in churning, does not increase the inner-race temperature because of the additional cooling by the greater flow.

Circumferential Temperature Distribution

The effect of the operating variables on the outer-race circumferential temperature gradient is presented in table II where the two methods of lubrication (single jet and oil puddling) are compared at a DN of 1.2×10^6. Although the temperatures were not uniform around the outer race of the bearing, the difference between the outer-race-maximum and outer-race-minimum temperatures did not exceed 14° F under any condition of operation and this difference occurred when the jet was set at location f where the oil jet was aimed at the face of the outer race (fig. 4). Outer-race-minimum temperature occurred in the region 0° to 120° after the oil-jet location in the direction of shaft rotation for single-jet lubrication.

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The exact effect of this circumferential gradient of the outer race is unknown and may be unimportant. However, it can cause the outer race to become out of round or cause thermal stresses in the outer race and housing when the gradient is high as in roller bearings (reference 1).

Effect of Bearing Geometry

One of the inherent features of ball bearings is the fact that the outer-race groove traps oil in the bearing and distributes it evenly around the outer race (oil may be observed to flow from the bearing around its complete circumference). This even distribution results in low circumferential temperature gradients.

Effect of Running Time on Bearing Operating Characteristics

The data presented in figure 16 show the effect of running time on the operating characteristics of ball bearings 19 and 20. The data were obtained at the reference conditions (DN, 1.2×10^6 (16,000 rpm); load, 368 lb; oil flow, 2.75 lb/min; oil inlet temperature, 100° F; and oil-jet diameter, 0.050 in.); the oil stream was directed at the space between the cage and the inner race. The running time of bearing 19 was 59.6 hours, and of bearing 20, 78.4 hours. The operating conditions listed were chosen because the effects of small changes of bearing dimensions, viscosity, jet location, and so forth are sufficiently magnified. The abscissa was arbitrarily selected in reference 1 and is used herein to represent, to a first approximation, the severity of bearing operation. This severity factor is the summation of the products of the difference between the outer-race-maximum temperature and the oil inlet temperature for each operating condition and the corresponding operating time in minutes at that particular condition.

The following data are presented in figure 16 for each bearing: outer-race-maximum and inner-race temperatures, oil viscosity at 100° F, and oil viscosity index.

The outer-race-maximum and the inner-race temperatures varied throughout the running period and virtually followed the same trend (fig. 16). The variations may be explained by small changes in operating conditions and lubricant properties. The over-all temperature effect of such small changes in viscosity, oil inlet temperature, oil flow, speed, jet location, and bearing geometry over the running period as obtained at the reference conditions was 8° F for the outer race and 16° F for the inner race of bearing 19, and 19° F for the outer race and 4° F for the inner race of bearing 20. For any particular run, these temperatures are reproducible to within ±3° F.

There was an appreciable decrease in lubricant viscosity with increasing operating severity; similar behavior is reported in reference 1 for roller-bearing operation.

Changes in Test-Bearing Dimensions with Running Time

The test-bearing measurements, taken before and after running, are given in table I. Inasmuch as nondestructive disassembly of the bearings investigated is impossible, unused sample bearings were disassembled to obtain the necessary initial measurements. Comparison of the before-and-after running data is open to question as the data compared were not obtained from the same bearing. However, since high-speed aircraft-grade bearings are manufactured to very close tolerances, it is possible to make the following observations from the data obtained:

(a) There was no significant change in ball diameters.

(b) There was negligible wear in the cage pocket and negligible change in clearance between the ball and the cage pocket.

(c) Bearing and cage diametral clearance did not increase any measurable amount.

(d) Bearing axial clearance increased by 0.004 inch in bearing 19, an amount which is not appreciable for axial play.

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Comparison of Single-Jet and Puddling Lubrication

Puddling produces lower outer-race temperatures, while single-jet lubrication produces lower inner-race temperatures (fig. 17). It is of interest to note how very close the outer-race temperature for puddling lubrication is to that obtained with single-jet lubrication at the same operating conditions (fig. 4). If the flow-ratio curve is extrapolated to zero (the ratio of flows for puddling lubrication), the outer-race temperature obtained is approximately 170° F. The outer-race temperature for puddling lubrication at the same operating conditions (DN, 1.2×10^6; load, 368 lb; oil flow, 2.75 lb/min) is also 170° F. Thus, the effectiveness of a given quantity of oil in cooling the outer race of a bearing under any given set of operating conditions is increased as the amount of oil transmitted through the bearing is increased. This is not true however, for the inner race; the inner-race temperature increases at first with oil flow because of oil churning in the bearing, but then decreases because of the cooling effect of the large volume of oil, in spite of the further increase in churning.

SUMMARY OF RESULTS

The following results were obtained in an experimental investigation of the effects of DN, load, oil flow, and oil inlet conditions on the operating characteristics of two 75-millimeter-bore (size 215) inner-race-riding cage-type ball bearings, which were operated over a range of DN values (bearing bore in mm times shaft speed in rpm) from 0.3×10^6 to 1.2×10^6, oil flows from 2 to 8 pounds per minute, and loads from 7 to 1113 pounds:

1. The radial location of the oil jet and the distribution of the oil were found to be important factors in the lubricating and cooling effectiveness of a given quantity of oil. Of the six jet positions investigated, lowest average bearing temperatures were obtained when the oil was directed exactly at the space between the cage and the inner race.

2. The effectiveness of lubrication, as indicated by outer-race-maximum temperature, depended on the quantity of the oil transmitted through the bearing regardless of the method of supplying oil. The inner-race temperature, however, was dependent on the method of oil supply, and did not always decrease with greater transmitted-oil flow.

3. Oil supplied to the bearing by puddling the oil through the bearing produced lower outer-race temperatures and higher inner-race temperatures than did oil supplied to the bearing by a single jet.

4. The horsepower rejected to the oil increased with increasing transmitted-oil flow at a constant total oil flow of 2.75 pounds per minute for single-jet lubrication and was maximum when all the oil was transmitted through the bearing (achieved by puddling).

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5. When all the oil was transmitted through the bearing (puddling lubrication), the oil outlet temperature and the outer-race-maximum temperature were approximately the same. These temperatures may be predicted from the data obtained when a single jet was located at several radial positions by extrapolating the curve of bearing temperature against the flow ratio to zero.

6. The ratio of deflected-oil flow to transmitted-oil flow increased with increasing DN at a load of 368 pounds; at oil flows higher than 4 pounds per minute, this flow ratio was nearly constant over the range of DN values investigated.

7. Bearing operating temperatures were most sensitive to changes in speed, and the outer-race-maximum temperature increased linearly with increasing DN. The inner-race temperature, however, increased with DN at an increasing rate.

8. Bearing operating temperatures increased with increasing load and decreased with increasing oil flow at DN values of 0.735×10^6 and higher. At low speed (DN of 0.3×10^6), neither load nor oil flow had any appreciable effect on bearing operating temperatures.

9. The outer-race circumferential temperature gradients of the test bearing were small, only 14° F under even the most adverse condition of lubrication (jet directed at the outer-race face of the bearing). In most cases, the circumferential gradient was about 7° F.

Lewis Flight Propulsion Laboratory
National Advisory Committee for Aeronautics
Cleveland, Ohio, August 27, 1952

REFERENCES

1. Macks, E. Fred, and Nemeth, Zolton N.: Investigation of 75-Millimeter-Bore Cylindrical Roller Bearings at High Speeds. I - Initial Studies. NACA TN 2128, 1950.

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2. Macks, E. Fred, and Nemeth, Zolton N.: Investigation of 75-Millimeter-Bore Cylindrical Roller Bearings at High Speeds. II - Lubrication Studies - Effect of Oil-Inlet Location, Angle, and Velocity for Single-Jet Lubrication. NACA TN 2216, 1950.

3. Macks, E. Fred, and Nemeth, Zolton N.: Lubrication and Cooling Studies of Cylindrical-Roller Bearings at High Speeds. NACA Rep. 1064, 1952. (Supersedes NACA TN 2420.)

4. Macks, E. Fred, Anderson, William J., and Nemeth, Zolton N.: Influence of Lubricant Viscosity on Operating Temperatures of 75-Millimeter-Bore Cylindrical-Roller Bearing at High Speeds. NACA TN 2636, 1952.

5. Getzlaff, Günter: Experiments on Ball and Roller Bearings under Conditions of High Speed and Small Oil Supply. NACA TM 945, 1940.

6. Boyd, John, and Eklund, P. R.: Some Performance Characteristics of Ball and Roller Bearings for Aviation Gas Turbines. Paper No. 51-A-78, Aviation Div., presented at A.S.M.E. Meeting, Atlantic City (N.J.), Nov. 25-30, 1951.

7. Dawson, J. G.: Lubricating Problems of the Gas Turbine Engine. Shell Aviation News, no. 133, July 1949, pp. 14-22.

8. Hunt, Kenneth C.: Petroleum Requirements of British Gas Turbines. II - Lubricants. SAE Jour., vol. 59, no. 11, Nov. 1951, pp. 20-21.

9. Wilcock, Donald F., and Jones, Frederick, C.: Improved High-Speed Roller Bearings. Lubrication Eng., vol. 5, no. 3, June 1949, pp. 129-133; discussion, vol. 5, no. 4, Aug. 1949, p. 184.

10. Tarr, Philip, R.: Methods for Connection to Revolving Thermocouples. NACA RM E50J23a, 1951.

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TABLE I - PHYSICAL CHARACTERISTICS OF TEST BEARINGS

Bearing number	19		20	
Construction	Two-piece inner-race-riding cage		Two-piece inner-race-riding cage	
Number of balls	11		11	
Pitch diameter of bearing, in.	4.035		4.035	
	Before	After	Before	After
Total running time, hr	0	59.6	0	78.4
[a]Severity factor	0	200,688	0	302,112
Ball diameter, in.	0.6872[b]	0.6871	0.6872[b]	0.6870
Diametral clearance between cage and ball, in.	0.0230	0.023	0.0230	Bearing run to failure in DN limiting test
Unmounted bearing:				
Diametral clearance, in.				
Bearing	0.0012	0.0012	0.0006	
Cage	.027	.028	.028	
Eccentricity	.0002	.0001	.00015	
Axial clearance, in.				
Bearing	.006	.010	.0058	
Mounted bearing:				
Diametral clearance, in.				
Bearing	0.0004	0.0003	0.0001	
Cage	.027	.028	.028	
Eccentricity	.0002	.0004	------	

[a]Summation of products of difference between outer-race-maximum temperature and lubricant inlet temperature for each operating condition and corresponding operating time in minutes at that particular condition.

[b]Measurements obtained from sample bearing.

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TABLE II - OUTER-RACE CIRCUMFERENTIAL TEMPERATURE GRADIENTS

[Data obtained at DN of 1.2×10^6 and oil inlet temperature of 100° F]

Method of lubrication	Outer-race-temperature gradient, ΔT, °F											
	Load (lb) (a)			Oil flow (lb/min) (b)			Jet location (c)					
	7	368	1113	2.0	2.75	7.5	a	b	c	d	e	f
Puddling oil through bearing	-	$6\frac{1}{2}$	-	8	$6\frac{1}{2}$	3	--	--	-	-	-	--
Single jet of 0.050-in. diam	8	7	7	6	7	7	10	$6\frac{1}{2}$	6	7	8	14

[a]Data obtained at oil flow of 2.75 lb/min.
[b]Data obtained at load of 368 lb.
[c]Data obtained at oil flow of 2.75 lb/min. and load of 368 lb.

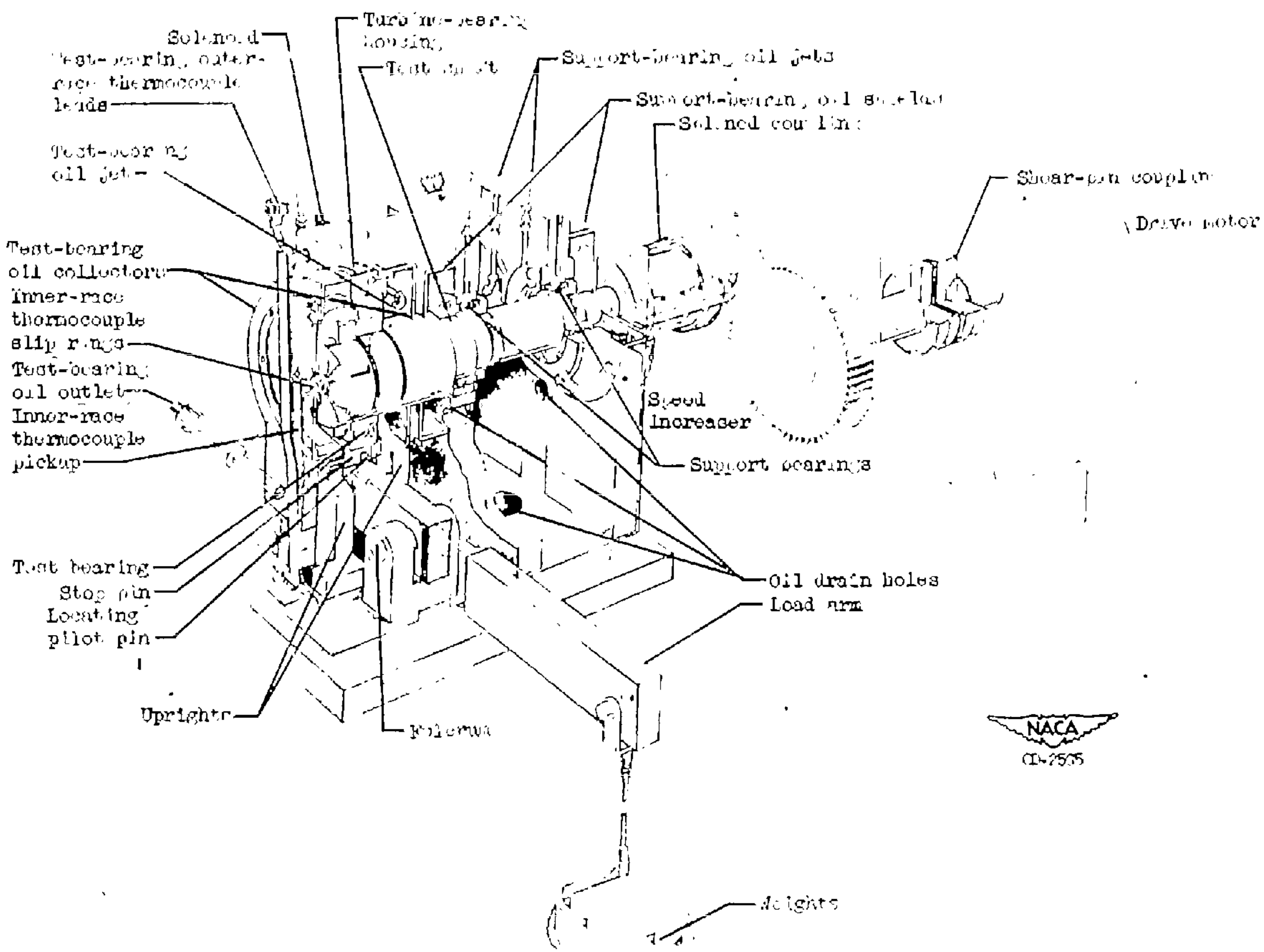

Figure 1. - Cutaway view of radial-load rig.

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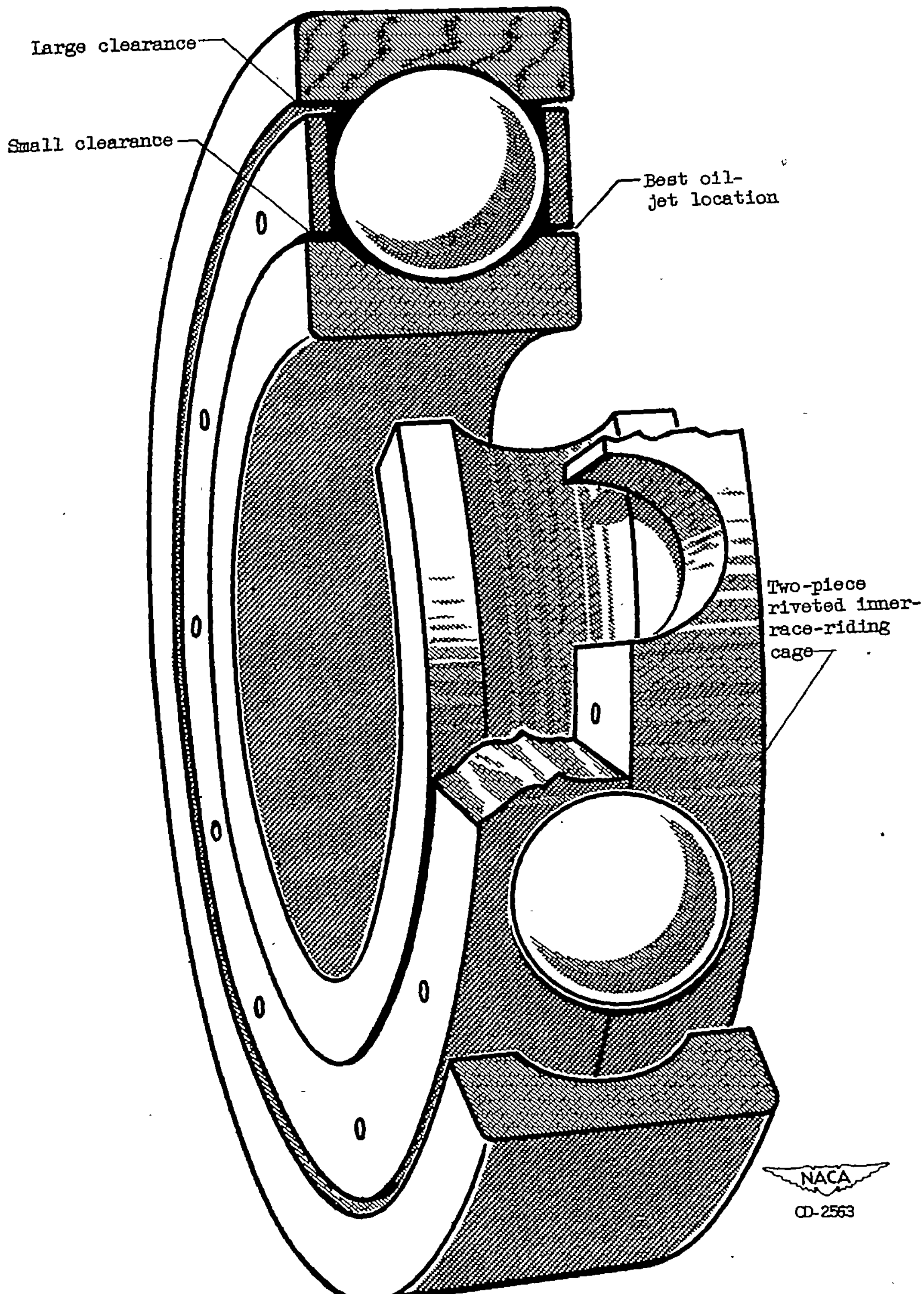

Figure 2. - Schematic diagram of test bearing.

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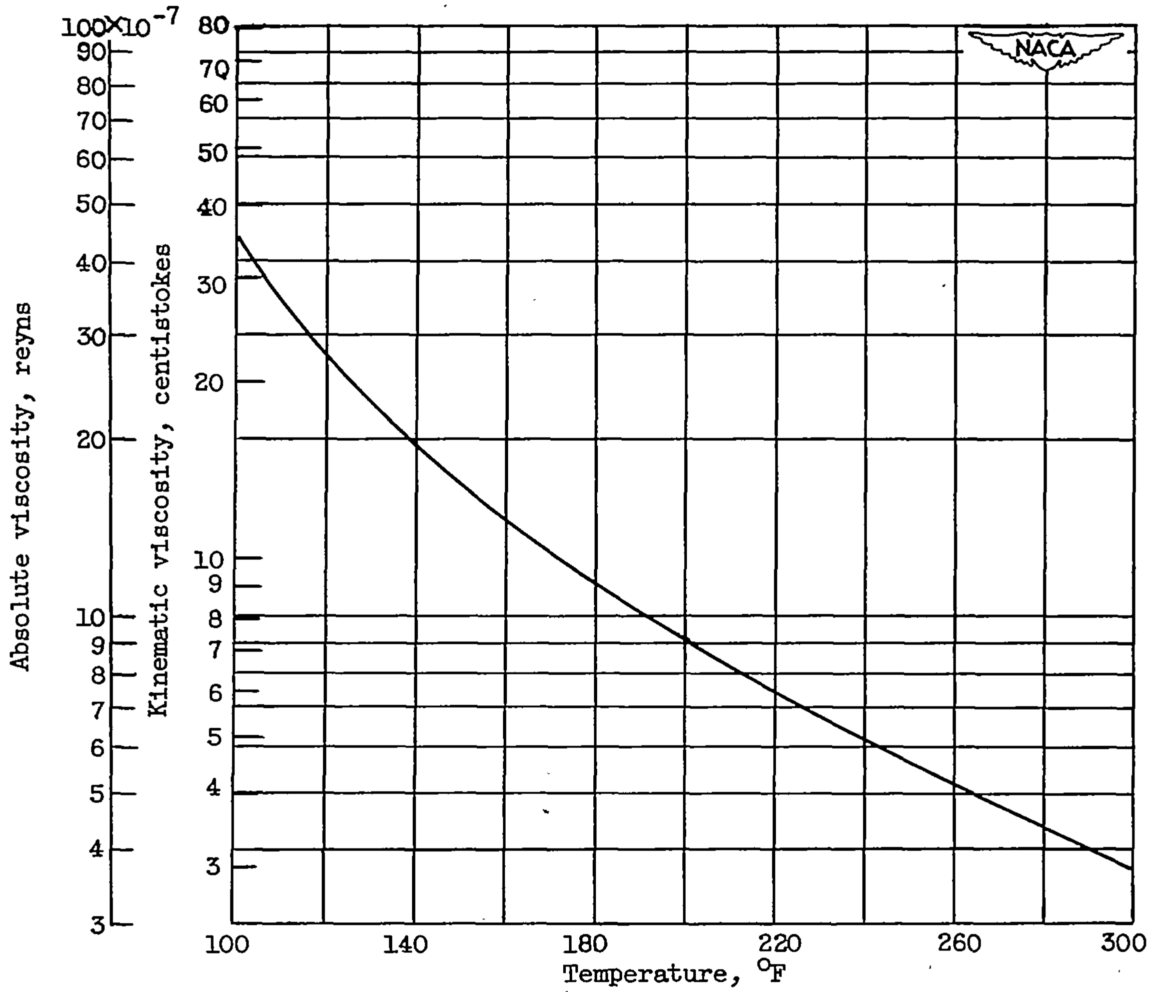

Figure 3. - Effect of temperature on kinematic and absolute viscosities of test oil. Pour point, -50° F; flash point, 310° F; viscosity index, 150; autogeneous ignition temperature, 490° F.

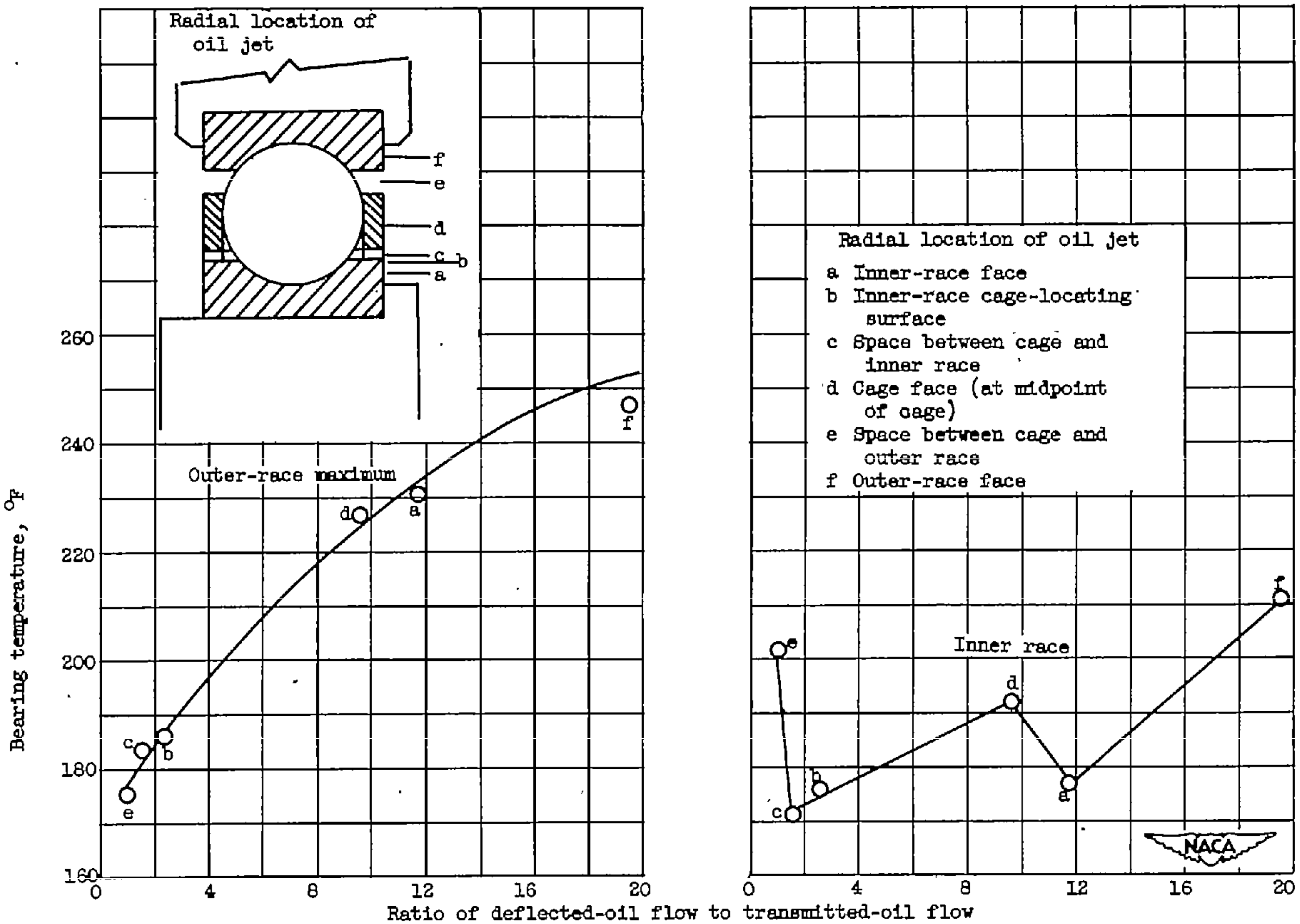

Figure 4. - Effect of ratio of deflected-oil flow to transmitted-oil flow on outer-race-maximum and inner-race temperatures of bearing 20. DN, 1.2×10^6; load, 368 pounds; oil flow, 2.75 pounds per minute; oil inlet temperature, 100° F; oil-jet diameter, 0.050 inch.

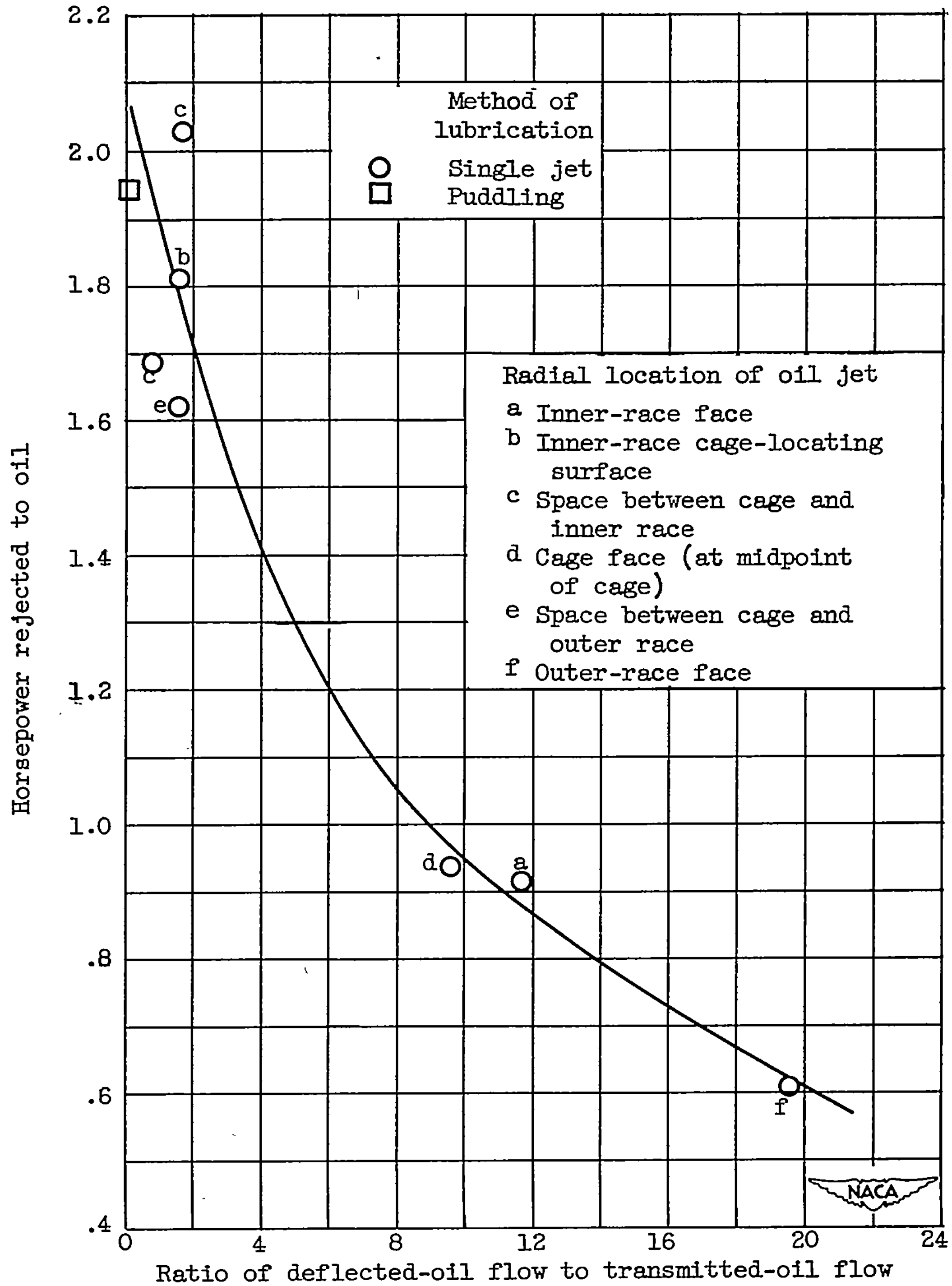

Figure 5. - Effect of ratio of deflected-oil flow to transmitted-oil flow on the horsepower rejected to the oil for bearing 20. DN, 1.2×10^6; load, 368 pounds; oil flow, 2.75 pounds; oil inlet temperature, 100° F.

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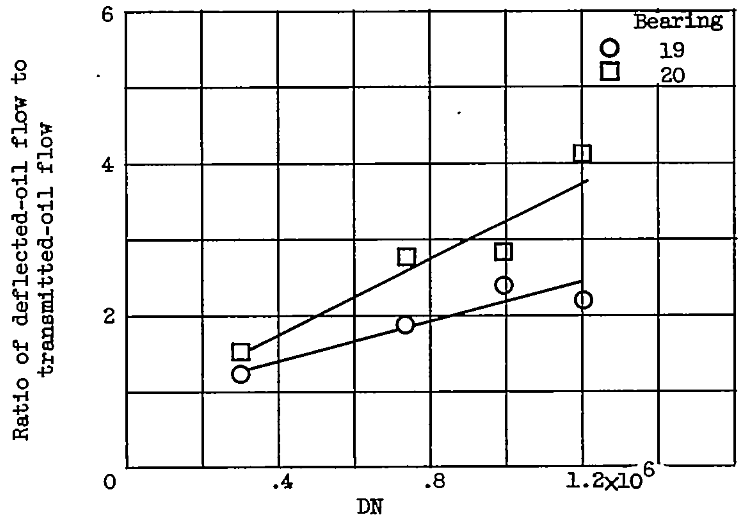

Figure 6. - Effect of DN on ratio of deflected-oil flow to transmitted-oil flow for bearings 19 and 20. Load, 368 pounds; oil flow, 2.75 pounds per minute; oil inlet temperature, 100° F; and oil-jet diameter, 0.050 inch.

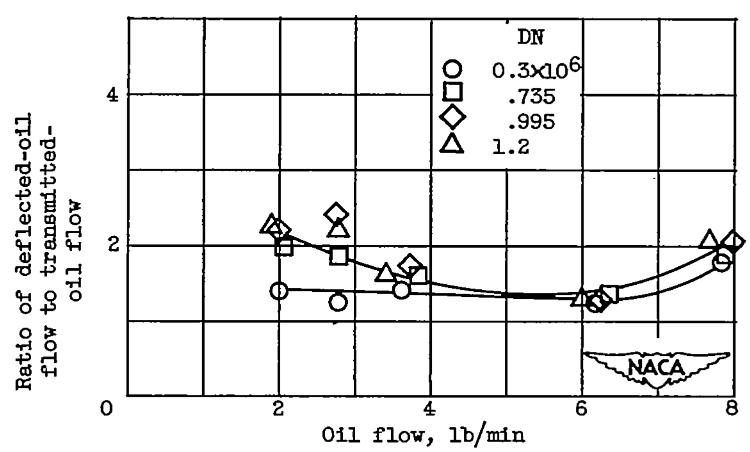

Figure 7. - Effect of oil flow on ratio of deflected-oil flow to transmitted-oil flow for bearing 19. DN, 0.3×10^6, 0.735×10^6, 0.995×10^6, and 1.2×10^6; load, 368 pounds; oil inlet temperature, 100° F, and oil-jet diameter, 0.050 inch.

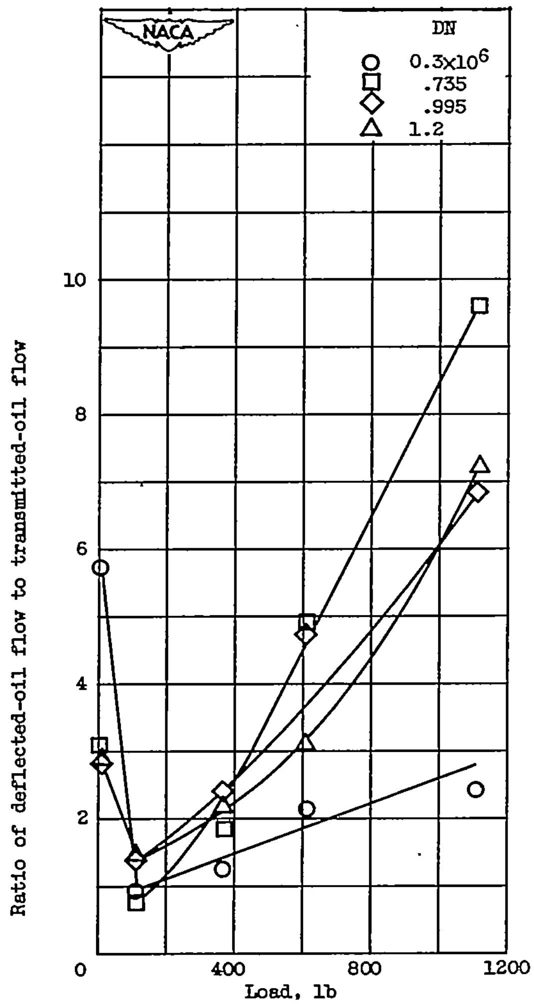

Figure 8. - Effect of load on ratio of deflected-oil flow to transmitted-oil flow for bearing 19. DN, 0.3×10^6, 0.735×10^6, 0.995×10^6, 1.2×10^6; oil flow, 2.75 pounds per minute; oil inlet temperature, 100° F, and oil-jet diameter, 0.050 inch.

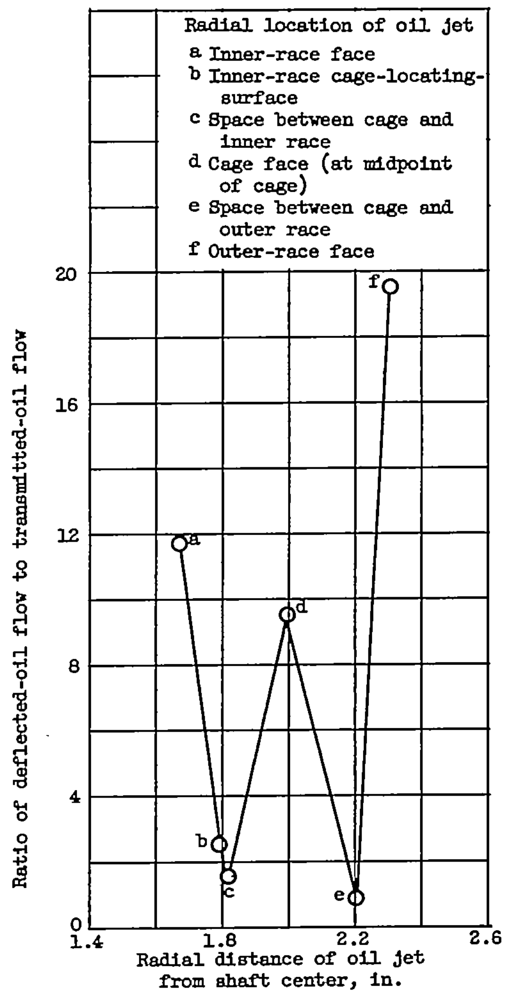

Figure 9. - Effect of radial location of oil jet on ratio of deflected-oil flow to transmitted-oil flow for bearing 20 with single-jet lubrication. DN, 1.2×10^6; load, 368 pounds; oil flow, 2.75 pounds per minute; oil inlet temperature, 100° F; oil-jet diameter, 0.050 inch.

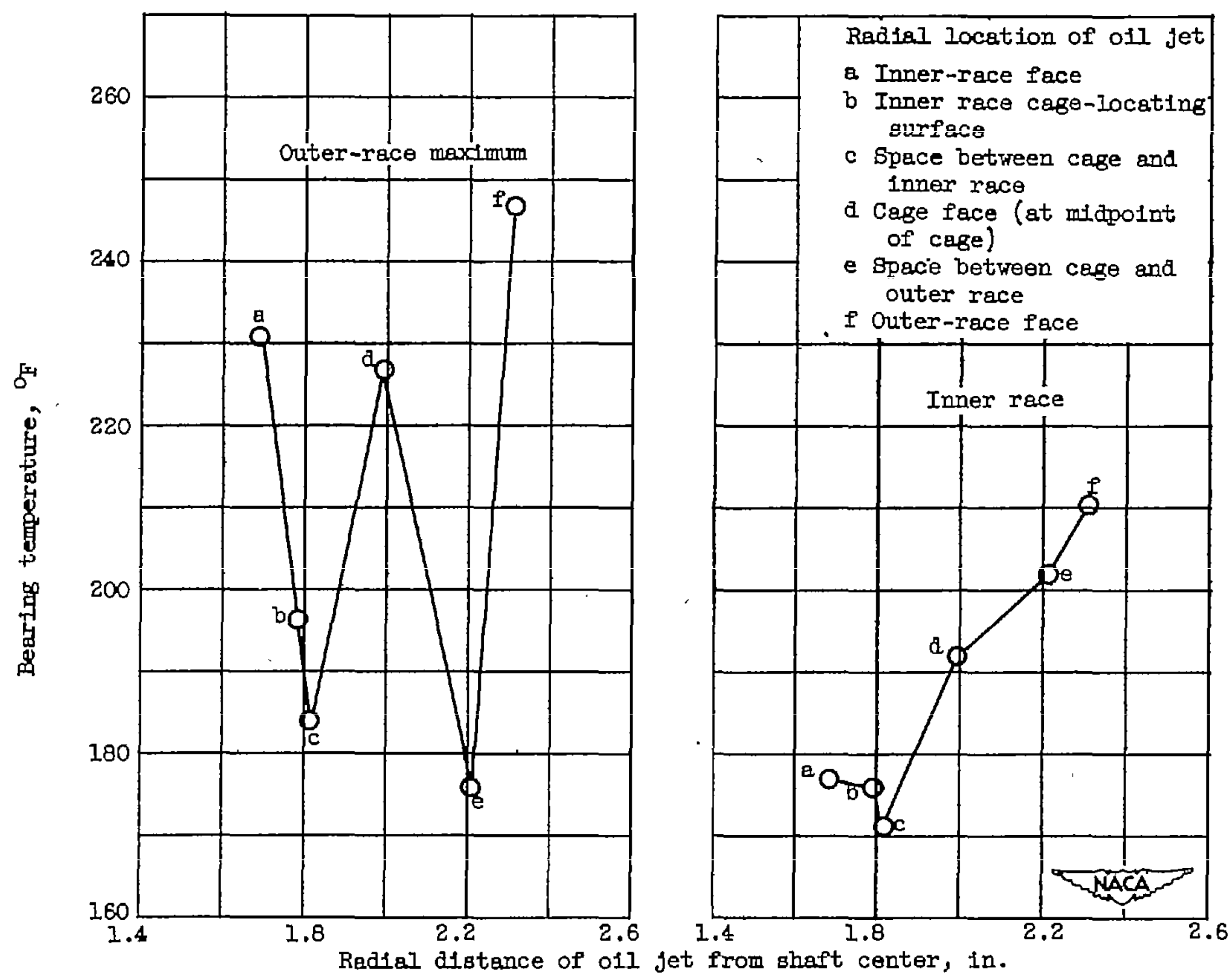

Figure 10. - Effect of radial location of oil jet on operating temperatures of bearing 20 with single-jet lubrication. DN, 1.2×10^6; load, 368 pounds; oil flow, 2.75 pounds per minute; oil inlet temperature, 100° F; oil-jet diameter, 0.050 inch.

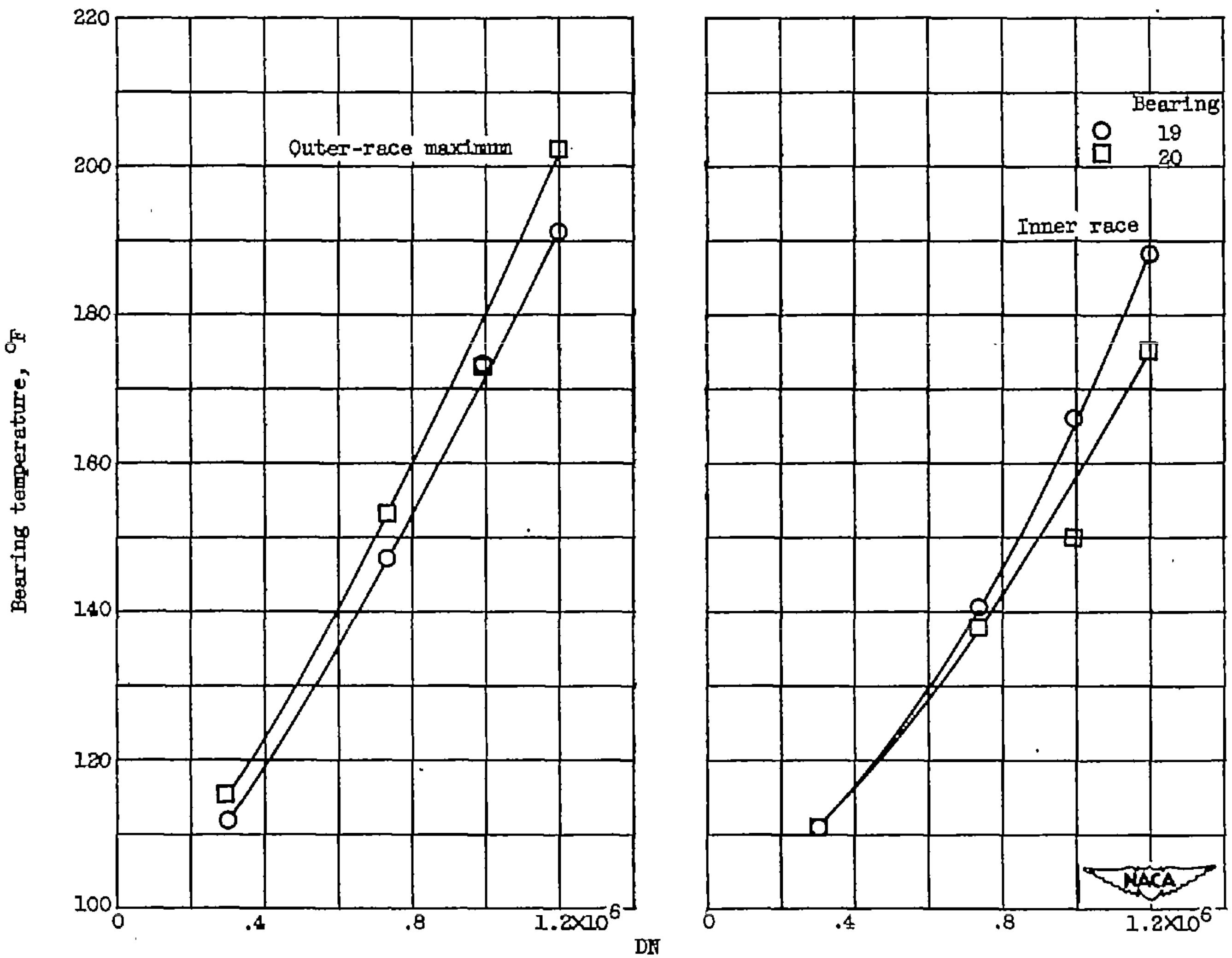

Figure 11. - Effect of DN on outer-race-maximum and inner-race temperatures of bearings 19 and 20. Load, 368 pounds; oil flow, 2.75 pounds per minute; oil inlet temperature, 100° F, and oil-jet diameter, 0.050 inch.

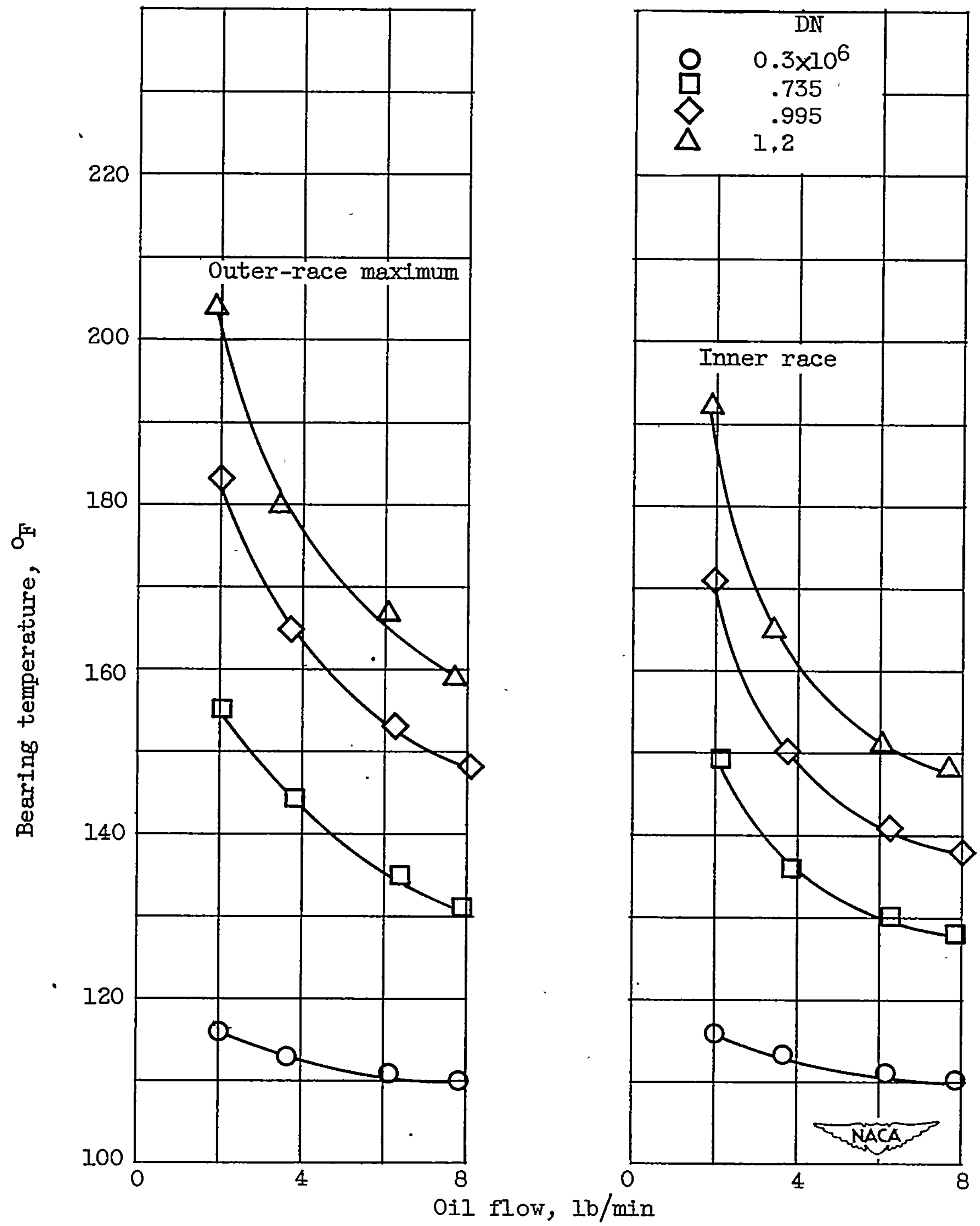

Figure 12. - Effect of oil flow on outer-race-maximum and inner-race temperatures of bearing 19. DN, 0.3×10^6, 0.735×10^6, 0.995×10^6, and 1.2×10^6; load, 368 pounds; oil inlet temperature, 100° F; oil-jet diameter, 0.050 inch.

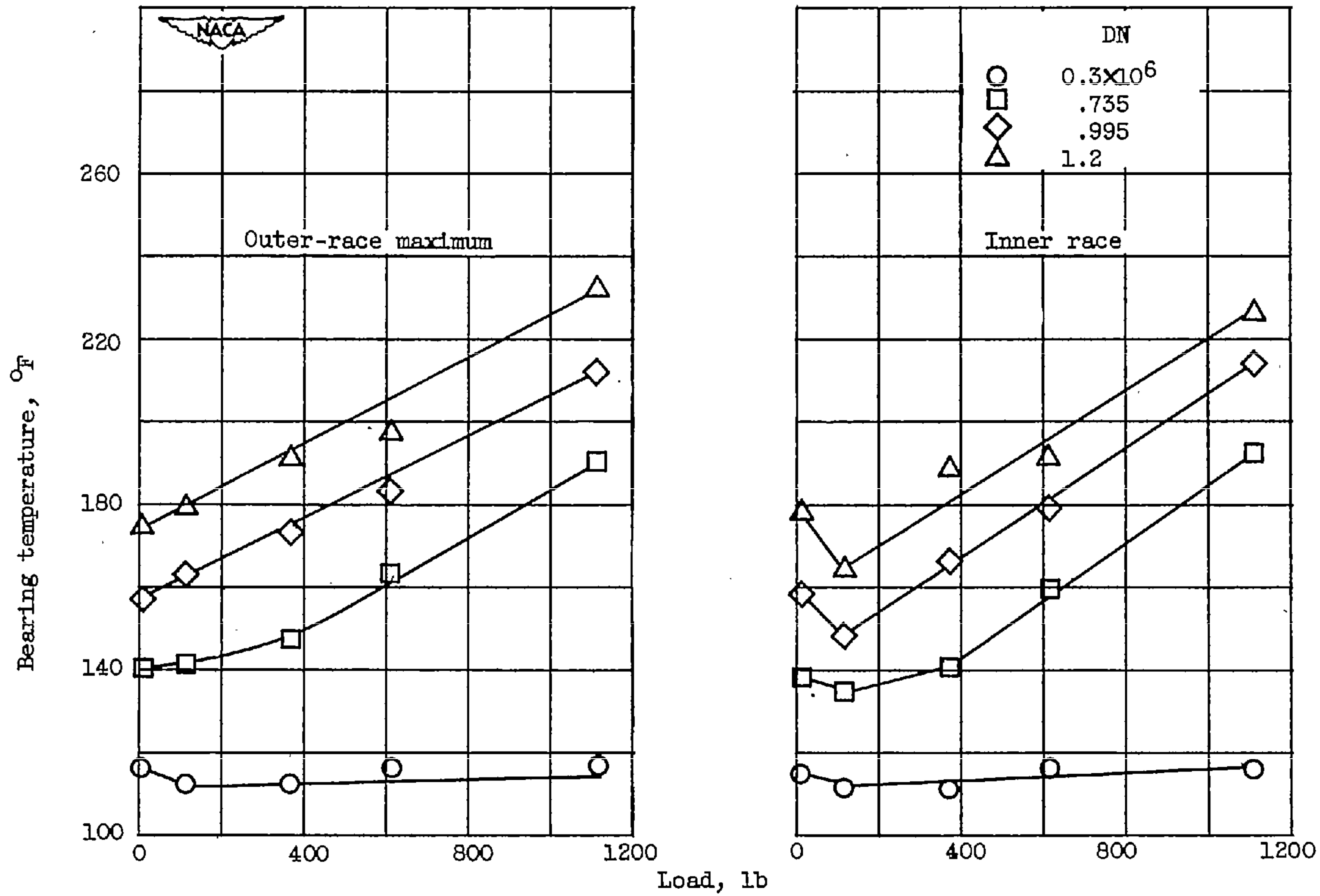

Figure 13. - Effect of load on outer-race-maximum and inner-race temperatures of bearing 19. DN, 0.3×10^6, 0.735×10^6, 0.995×10^6, 1.2×10^6; oil flow, 2.75 pounds per minute; oil inlet temperature, 100° F; oil-jet diameter, 0.050 inch.

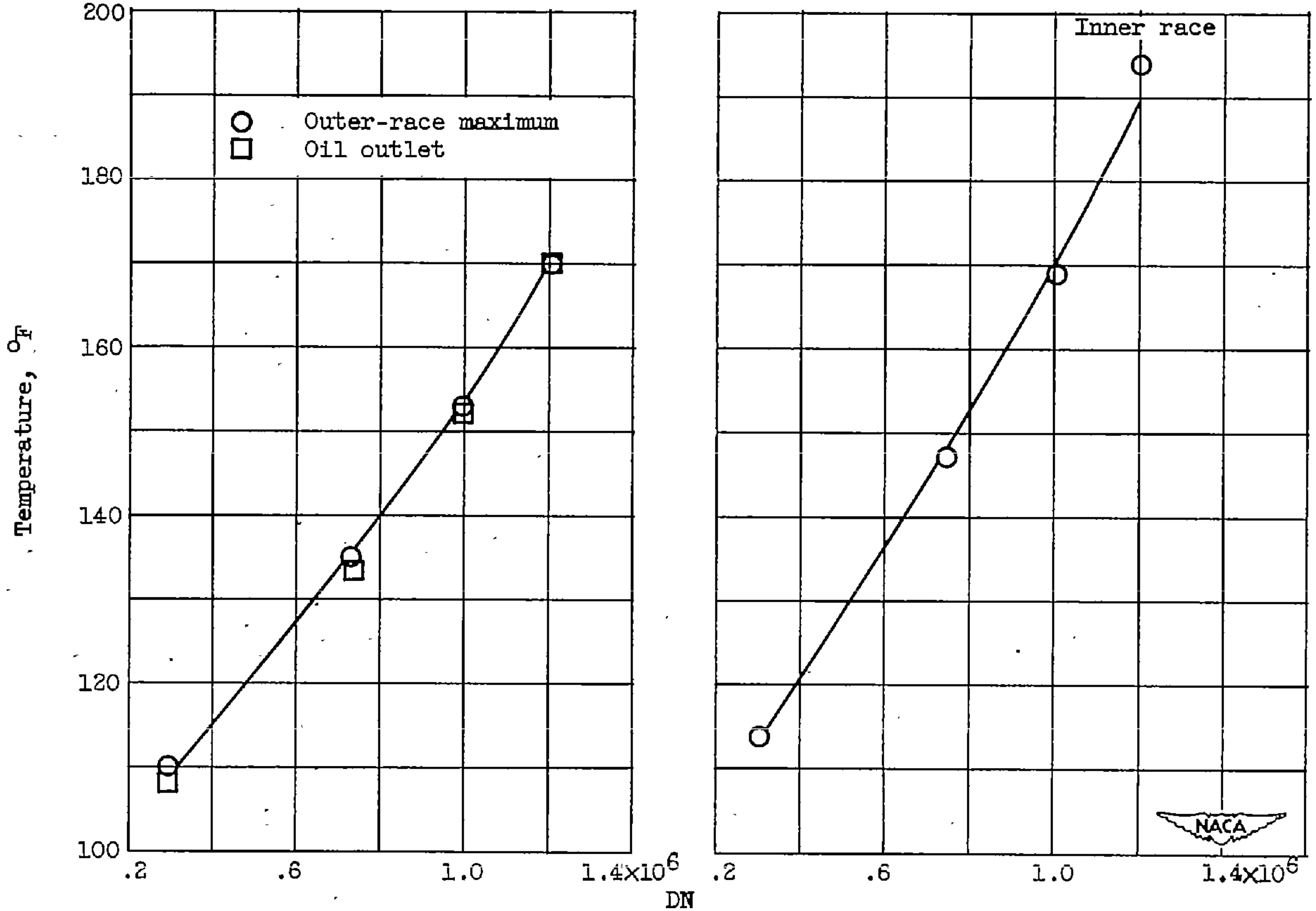

Figure 14. - Effect of DN on outer-race-maximum, inner-race, and oil outlet temperatures for oil puddled through bearing 20. Load, 368 pounds; oil flow, 2.75 pounds per minute; oil inlet temperature, 100° F.

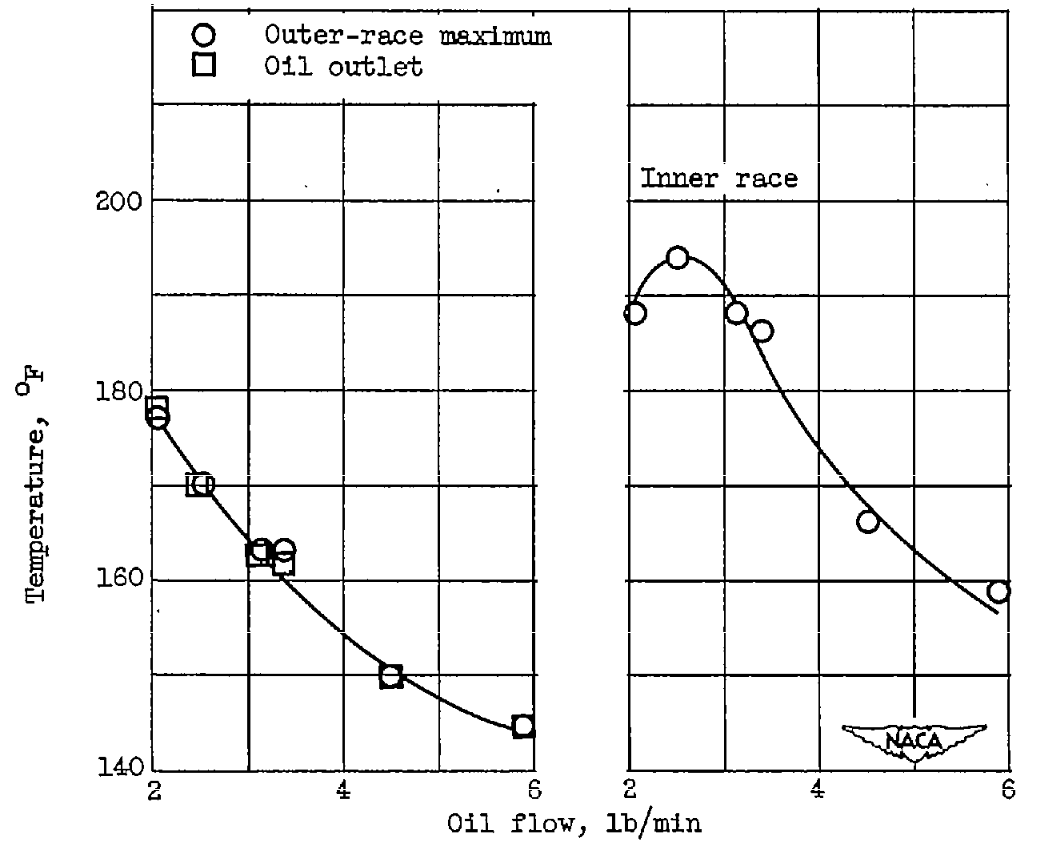

Figure 15. - Effect of oil flow on outer-race-maximum, inner-race and oil outlet temperatures for oil puddled through bearing 20. DN, 1.2×10^6; load, 368 pounds; oil inlet temperature, 100° F.

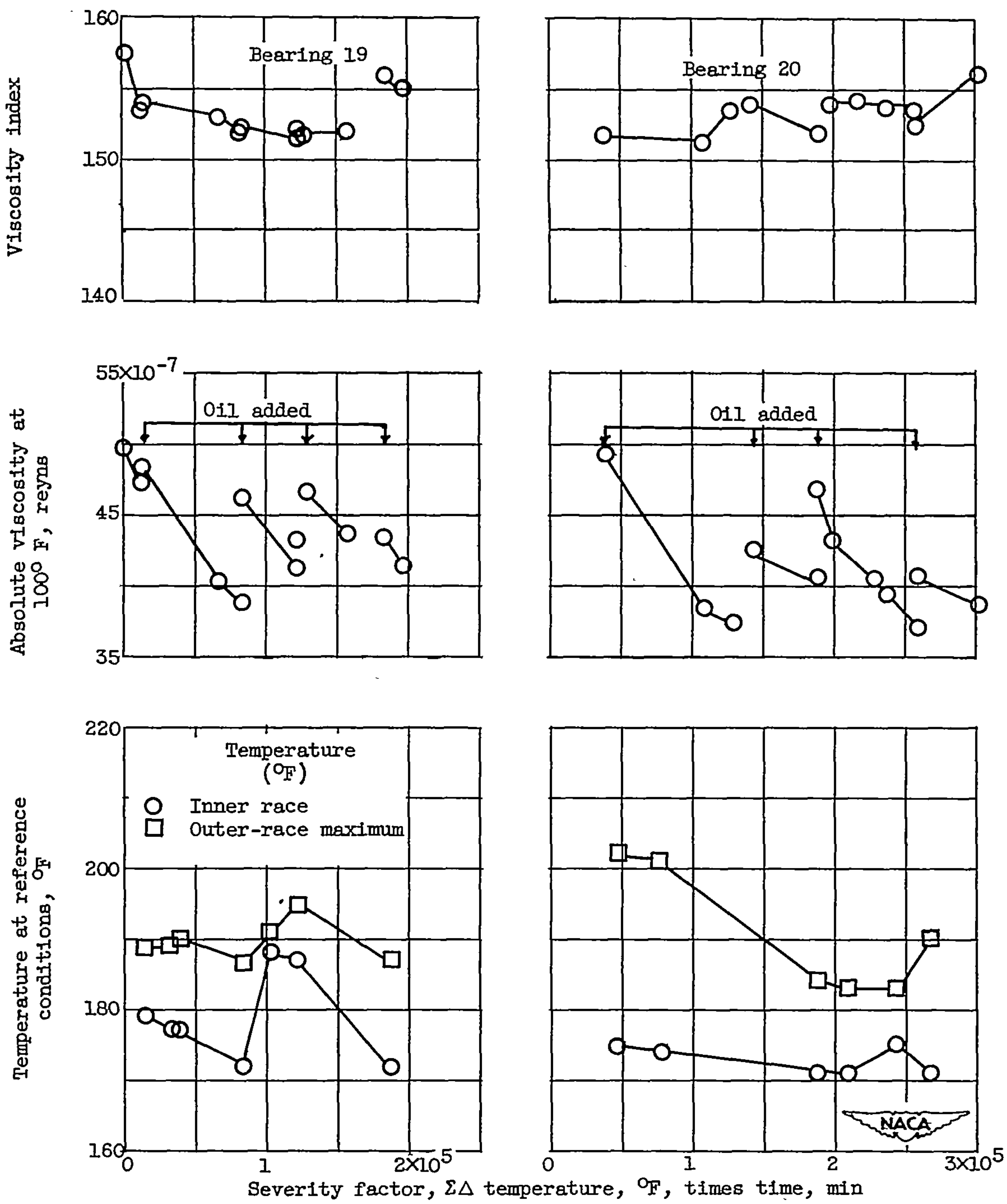

Figure 16. - Effect of severity factor on bearing operating characteristics and lubricant properties for bearings 19 and 20. DN, 1.2×10^6; load, 368 pounds; oil flow, 2.75 pounds per minute, oil inlet temperature, 100° F; oil-jet diameter, 0.050 inch. Oil stream directed at space between cage and inner race.

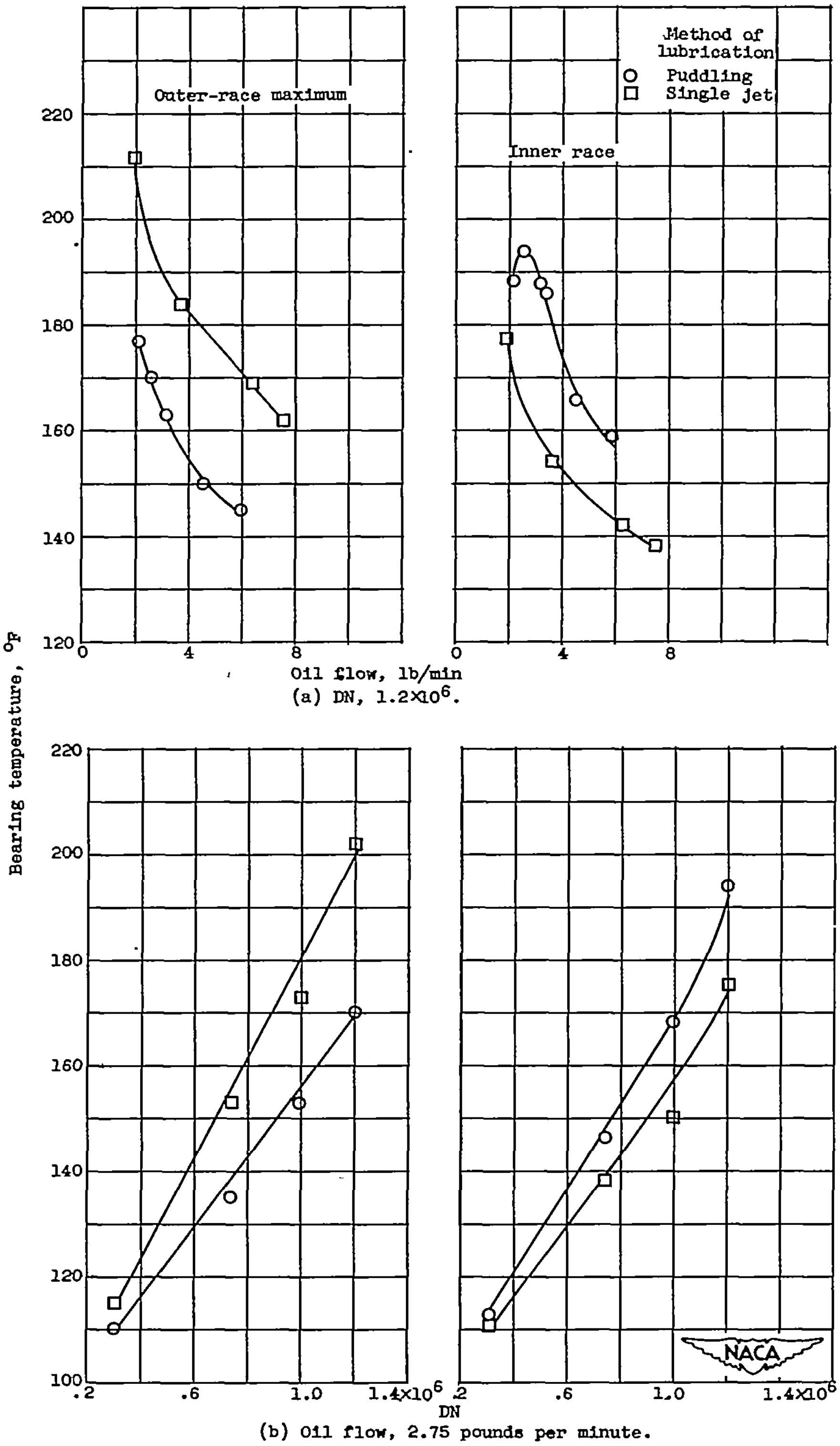

Figure 17. - Effect of oil flow and DN on outer-race-maximum and inner-race temperatures of bearing 20 for two methods of lubrication. Methods of lubrication, single jet and puddling. Load, 368 pounds; oil inlet temperature, 100° F; oil-jet diameter, 0.050 inch.